SPECIFICATION

Docket No.: 20639.001

TO ALL WHOM IT MAY CONCERN:

BE IT KNOWN that I, R. David Anderson, a citizen of the United States of America, residing in the State of Texas, have invented new and useful improvements in a

THERMAL ENERGY STORAGE DEVICE AND METHOD

of which the following is a specification:

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BACKGROUND OF THE INVENTION

1. Field of the Invention:

The present invention relates to an energy storage device that can easily be added to an existing Freon compression air-conditioning system commonly used in residential and small commercial applications. The method of the invention allows the addition of a single thermal energy storage unit which will replicate the operation of a conventional condensing unit while using only energy that was previously stored in the existing system without any change or alteration of existing equipment.

2. Description of the Prior Art:

As a result of the growing popularity of air-conditioning, electrical demands have increased drastically in certain areas. A particular problem exists because of the common demand for air-conditioning power during the hottest portion of the day. Such demands tax electrical generation plants and the electrical distribution systems presently in place. In order to counter this common demand, electric utilities currently offer incentives including significantly reduced rates for electricity used during off peak hours.

For example, it has become common for the consumer to be offered a two-tiered electrical pricing structure. Available incentives for the consumer include a lower price for electricity when used at low demand times (off peak time) and a higher price for high demand time use (peak time). These incentives are an effort to more efficiently utilize existing electrical facilities (power plants, power lines, etc.) and to defer the need for additional facilities and equipment to provide the capacity to meet the peak time demand. Significantly, the peak time demand hours currently represent only about 8% of the total available hours in a year.

As a typical example, a consumer who has been paying a uniform ten cents per kilowatt hour of electrical use might only pay three cents per kilowatt hour for the same amount of power during off peak times while paying fifteen cents during peak times. The definition of "peak time" varies according to a number of variables including the geographic location, the particular generation stations involved, the distribution methods utilized, the local population etc. Generally, the peak time hours are from noon until sundown during the summer months with other times being defined as off

peak. As should be apparent from the foregoing discussion, it has become economically advantageous for the consumer to buy and store energy during the off peak times and to use the stored energy during peak times.

In order to take advantage of existing incentives offered by the electrical utilities, larger commercial users of electricity have found it feasible to shift their electric use to off peak hours by using any of several different methods. These existing methods have included: (1) using multiple air conditioners and scheduling building use such that all of the space cooling equipment does not operate at the same time; (2) shifting the operation schedule of equipment that can be operated during off-peak hours by operating such equipment on a timer without supervision where the operation requires no longer than 16 hours to complete its process; (3) changing the operating hours of employees to off-peak hours, including offering shift premiums for employees willing to cooperate; and (4) installing a thermal storage system for space cooling needs which stores energy during off-peak hours and then utilizes the stored energy during peak hours.

The thermal storage method listed as (4) above is particularly pertinent to the improvement offered by the method and device of the present invention. The existing thermal storage systems include:

1. Cooling large volumes of water during off-peak hours and then circulating the cooled water through the structure to be cooled during peak hours.

2. Freezing a smaller volume of water to ice, followed by circulating a liquid through the ice for cooling and by then flowing the liquid inside the structure where it cools the air and then returns to the ice bank to once again be cooled. Such systems take advantage of the greater energy storing capacity which occurs with the phase change of water from a liquid to solid. Using water as the storage media, 144 BTU of heat can be transferred to the ice per pound before the ice completely melts.

3. Freezing a volume of water to ice where the ice is located in a tank which surrounds a coil containing a refrigerant (typically R-22, Freon) used in the conventional cooling process during the off-peak hours with a conventional condensing unit. During peak hours the Freon refrigerant

condenses to near 100% liquid in the ice tank and is then is circulated with conventional pumping equipment to an evaporative coil located inside the structure. The Freon in the evaporative coil removes heat from the structure by evaporating the Freon to near its gaseous state. Because the pressure in the evaporative coil is only slightly higher than in the condensing coil of the ice tank (the difference is the pressure drop in the line and any elevation difference), the vaporizing temperature of the Freon in the evaporative coil is only slightly higher than the condensing temperature in the ice tank condensing coil. By this means, heat is transferred from inside the structure to the ice tank until the ice is melted.

Thermal storage devices of this last type have had some success in new construction, but have had very limited use in existing structures because of the initial expense and inconvenience of installation. Approximately 85% of the presently existing office space in this country was constructed prior to 1990. As a result, for any significant shift in the amount of electrical demand to off-peak hours to occur, a thermal storage system must be provided which is not only low in initial cost but which can also be installed with a minimum of effort and inconvenience. One limitation on the use of existing technology has been the requirement that the existing equipment be greatly modified, including modification of the evaporative coil or related equipment already located within the interior of the structure being cooled.

Another principal limitation on the use of the third type of thermal energy storage system described above relates to the various difficulties that are encountered in pumping liquid Freon which is condensed in the ice tank. An important part of any thermal energy storage device is the ability to cool the structure in the event the energy storage medium is depleted. With the first two types of systems described above, this task is easily accomplished because there are two coils in the storage medium. One of the coils is for cooling the medium with conventional methods while the other coil contains the coolant that is circulated through the structure. In the event of energy depletion of the storage medium, it is an easy matter to cool the energy storage medium with conventional means, which in turn cools the circulating fluid.

With the third type of device there is one coil in the storage medium which serves multiple purposes. In one time frame a conventional condensing unit is employed for (1) freezing the storage medium

with compression of the refrigerant vapor; (2) condensing the vapor to its liquid state in a condensing coil; (3) holding back pressure on the Freon with an expansion device so that condensing can take place at the ambient temperature of the condensing coil; and (4) freezing the storage medium after expansion of the liquid to the suction pressure of the compressor inside the single coil in the storage medium.

In a second time frame, the coil in the storage medium is used to condense the Freon returning from the structure. After the Freon is condensed to near 100% liquid, it is then pumped to an evaporative coil inside the structure. The refrigerant is vaporized in the evaporative coil due to the heat absorbed inside the evaporative coil and is then returned to the storage medium coil for condensing. It is then again pumped into the structure. An expansion device is not needed in this second time period because both the condensing coil and the evaporative coil operate at essentially the same pressure.

A third time period occurs in the case of the energy storage being depleted or cooling being needed in the structure during off-peak periods. During this third time period many different techniques have been employed to cool the structure. A basic problem exists with respect to all of the existing methods, however, because cooling of the structure with a conventional condensing unit requires an expansion device to hold back pressure on the cooling coil to enable the hot high pressure Freon leaving the compressor to condense to a liquid at a temperature that is close to ambient. This back pressure for R-22 is in excess of 200 psig, causing a differential pressure of over 100 psig between the suction and discharge of the compressor. This pressure differential is easily provided by the compressor but has not been achieved by the liquid pump required in the second time period because of the characteristic properties of the liquid Freon present in the ice tank summarized above. The following patents represent the current state of the art in attempting to solve this problem:

In United States Patent No. 4,735,064, issued April 5, 1988, to Fischer, the approach described moved the expansion device outside the structure in the vicinity of the ice tank where it served a dual purpose. In the first time period, the expansion device is used to make ice. In the third time period, the expansion device holds back pressure on the conventional condensing coil and allows the liquid to flash to low pressure before the evaporative coil inside the structure. Because the expansion device is located outside the structure, this method requires a new large and insulated line to be run

inside the structure to the evaporative coil. This is costly, inconvenient, and impractical in most cases.

In United States Patent No. 5,21 1,029, to Dean et al., issued May 18, 1993, the approach was to leave the expansion device inside the structure near the evaporative coil and to bypass the device during time period two operations. This arrangement works adequately in the time period three, but requires an additional line to be run into the structure for time period two operation when the pump is running because the pump can't overcome the high pressure which is present. Again this new line is costly, inconvenient, and impractical in most cases. This patent does contain a very brief comment that the pump could be discharged upstream of the expansion device, but it does not present any plan for getting around the pumping problems.

United States Patent No. 4,916,916, to Fischer, issued April 17, 1990 substitutes two volume tanks described as self pumpers for the liquid pump. This method requires the compressor to run during the second time period using the vapor high pressure discharge from the compressor to pressurize first one and then the other volume tank, thereby building up enough pressure to force the liquid Freon condensed by the ice tank inside the structure and through the expansion device. To save energy the patent proposes to replace the conventional compressor with a two-speed compressor. One speed would be used for time periods one and three and a reduced speed would be used for time period two. This method is costly, inconvenient, impractical but does demonstrate the problems involved in pumping the liquid refrigerant.

United States Patent No. 5,255,526, to Fischer, issued October 26, 1993, once again utilizes a liquid pump, leaves the expansion device inside the structure near the evaporative coil, and adds a refrigerant storage tank inside the structure. A bypass around the expansion device is provided inside the storage tank. This method requires a new line to be run from the pump into the structure to the storage tank and the installation of a storage tank inside the structure.

United States Patent No. 5,647,225, to Fischer, issued July 15, 1997, uses a liquid pump and adds an additional evaporator without an expansion device inside the structure. This new evaporator is

used in conjunction with the pump during time period two. This method requires that two new lines be run into the structure and the addition of a second evaporative coil.

United States Patent No. 5,467,812 , issued Nov. 21, 1995, to Dean et al., is a means of both heating and cooling a structure with stored energy. For the cooling phase a second evaporative coil is added inside the structure, without an expansion device, to be used with a refrigerant pump. This requires two new lines to be run into the structure and the addition of another evaporative coil.

United States Patent No. 5,678,626, issued October 21, 1997, to Dean, is similar in scope to the previous Dean patent with the additional benefit of simultaneous cooling with both evaporative coils. This requires that both the pump and the compressor be running at the same time, however. Since there are two evaporative coils now inside the structure, they both can be used at the same time. However, the original coil requires the compressor to be running. The time period in which both coils are likely to be required to operate simultaneously will most probably be the time of the highest electrical demand. However, from an economic view point, this is precisely the time that the compressor should not be running. This solution also requires two new lines to be run into the structure and the addition of another evaporative coil.

United States Patent No. 5,682,752, issued November 4, 1997, to Dean, abandons the concept of adding a second evaporative coil and describes a refrigerant pump that pumps liquid Freon into an expansion device. That device is described as a conventional thermal expansion valve having a sensor for appropriately controlling the flow of refrigerant to the evaporator. This type of expansion device is rarely used in residential and small commercial systems because of the attendant cost. As a result, it must be added in most installations. The problems of pumping the low viscosity liquid refrigerant are not discussed even though a gear pump is cited as the preferred type of pump to be used. The patent also teaches that proportional-integral control loop is required to insure proper refrigerant flow when using the added thermal expansion valve. This computer controlled proportional integral loop adjusts the pump speed as a function of the refrigerant temperature in the suction line. As slippage increases, the pump speed increases.

The problem of transferring refrigerant to the suction of the pump to raise the net positive suction pressure of the liquid in an effort to reduce cavitation is not addressed. The '752 patent does go on to describe a storage module, a first transitory mode, a second transitory mode, and accompanying computer logic. This described method also includes the use of a refrigerant storage module commonly known as a "liquid receiver" in conjunction with a pump down cycle. Such a cycle and receiver are described in a number of presently available refrigeration textbooks, e.g., "Heating and Cooling Essentials", by Killinger, The Goodheart-Willcox Company Inc., 1993 & 1999; "Modern Refrigeration and Air Conditioning" by Althouse, Turnquist, and Bracciano, The Goodheart-Willcox Company Inc., 1996, etc. By computer controlling the speed of the pump to compensate for slippage and by changing the expansion device from the common metering orifice/capillary tube type of expansion device to a thermally controlled expansion valve, a workable system may result. However, the system would be costly and have limited pump life.

In spite of the above advances in thermal energy storage systems, a need exists for a workable system which is simple in design and economical to install and which can be installed outside the existing structure to be cooled without the necessity of altering the existing equipment inside the structure.

SUMMARY OF THE INVENTION

The present invention provides a novel energy storage device or system which can easily be added by retrofit to an existing Freon compression air-conditioning system of the type commonly used in residential and small commercial structures. The system of the invention is also easily incorporated into new construction with relatively little additional expense. The method and device of the invention allow the addition of a single "package" to a new or existing air conditioning system that totally simulates the operation of the conventional condensing unit. The system of the invention uses only energy that was previously stored by the system and does not require any change or alteration of existing equipment inside the structure. The add-on package of the invention creates a new system that stores energy in a first time period, utilizes this stored energy during a second time period to cool the structure, and possesses the capacity to be able to cool the structure in a third time period without utilizing stored energy.

In addition to providing an add-on package which can be easily installed outside the structure, two distinct problems are solved by the improved system of the invention. The problem of pumping a low viscosity and easily vaporized liquid such as Freon is solved. Additionally, the problem of controlling the Freon flow and charge in each time period is solved without dependence upon the particular type of system which existed initially and was retrofitted.

The air conditioning system of the invention includes a compressor for compressing a refrigerant, the refrigerant being a compressible phase change fluid and a condensing unit operatively connected to the compressor. An evaporator unit and an associated expansion means are operatively interconnected to the condensing unit and to the compressor, the evaporator unit being in heat exchange relationship with a supply air stream for an indoor space inside a structure. The compressor is operable to circulate the refrigerant between the condensing unit and the evaporator unit to cool the supply air stream.

The air conditioning system of the invention is also comprised of a thermal energy storage unit including a tank having a thermal energy storage medium disposed therein and having an associated

heat exchanger. The heat exchanger is operably connected to the compressor and evaporator. The thermal energy storage unit may further including a temporary refrigerant storage tank.

A refrigerant circulating device is provided for circulating refrigerant through the heat exchanger in the tank and between the tank and the condenser and evaporator. Preferably, the refrigerant circulating device includes a prime mover and an auxiliary liquid which is acted upon by the prime mover, the auxiliary liquid being coupled to the refrigerant, whereby force exerted by the prime mover on the auxiliary liquid is indirectly transferred to the refrigerant.

A valve system is also provided for controlling the flow of refrigerant through the air conditioning system. The valve system is operative to provide three distinct time periods of operation for the system. A first time period allows refrigerant to flow from the condenser to the heat exchanger of the thermal energy storage unit to freeze the medium in the tank and to then return to the condenser without utilizing the evaporator. The second time period bypasses the condenser and circulates refrigerant through the thermal storage unit and through the evaporator to thereby cool the supply air inside the structure before returning to the thermal storage unit. The third time period utilizes only the temporary refrigerant storage vessel of the thermal storage unit and utilizes the condenser and evaporator of the air conditioning system to cool the supply air inside the structure as if the thermal storage unit were not present.

In a particularly preferred system of the invention, the prime mover is a positive displacement pump which communicates with a pair of fluid cylinders containing oil as an auxiliary fluid. The prime mover exerts a motive power upon pistons located within the fluid cylinders to thereby mechanically couple the motive power of the prime mover to the refrigerant being circulated in the system. The auxiliary liquid chosen, whether oil or another liquid, has a higher relative viscosity and a lower relative vapor pressure than the conventional Freon refrigerant. The preferred prime mover of the invention can be powered by a direct current motor and battery.

Additional objects, features and advantages will be apparent in the written description which follows.

BRIEF DESCRIPTION OF THE DRAWINGS

Figure 1 is a simplified schematic diagram representing the working of a gear type, positive displacement pump.

Figure 2 is a simplified schematic diagram which illustrates the problem of pumping liquid refrigerant in an air conditioning system of the type under consideration.

Figure 3 is a simplified schematic diagram of a conventional air conditioning system illustrating the inside and outside components thereof.

Figure 4 is a simplified schematic, similar to Figure 3, of another conventional type air conditioning system.

Figure 5 is a simplified schematic diagram of a pump unit used in the method of the invention.

Figure 6 is another embodiment of the pump unit which is used in the method of the invention.

Figure 7 is a simplified schematic diagram of one embodiment of the device and method of the invention.

Figure 8 is a simplified diagram, similar to Figure 7, showing another embodiment of the device and method of the invention.

Figure 9 is a simplified diagram, similar to Figure 8, showing the system of the invention installed within the inside and outside equipment of a conventional air conditioning system.

DETAILED DESCRIPTION OF THE INVENTION

The improvements represented by the present invention can perhaps best be understood if two of the principal problems to be overcome in the third type of thermal energy storage system described above are first explained. These problems relate to (1) the nature of the Freon refrigerant being pumped; and (2) flow control problems related to pumping the Freon refrigerant. These problems will be discussed in turn.

Problems With Pumping Freon:

Turning first to Figure 2, the problems associated with pumping condensed Freon are illustrated in schematic fashion. Coil 11 in Figure 2 is intended to represent a coil located within an ice tank and coil 14 is the conventional evaporative coil located inside the structure to be cooled. Because the coils 11 and 14 communicate through top conduit 13, they are both at the same approximate pressure. If the condensing coil 11 is made much larger than the evaporative coil 14, the pressure of the two would be controlled by the ice tank condensing coil and would be the condensing pressure at the temperature of the Freon in the ice tank. However, if the evaporative coil was made much larger than the condensing coil, the pressure of the two would be controlled by the evaporative coil and would be the evaporative pressure at the evaporative Freon temperature. In reality, this pressure is somewhere in between the two depending upon such factors as the flow rate and temperature of the air across the evaporative coil, the size of the two coils, and the heat transfer rate from the condensing coil into the ice tank.

After the system reaches equilibrium, the Freon in the condensing coil 11 is at saturation with some but not all of the Freon in the liquid form. If the pump 12 and the expansion device 15 were eliminated and if pressure drops associated with line friction were eliminated as well as any elevation differences, a natural circulation would develop in the system. In reality this doesn't occur because of pressure drops and elevation differences. In a well designed system, designed for this type of operation, it is possible to pump the liquid because there is very little pressure drop and the elevation differences can be minimized. In such a situation, the pump is merely required to develop enough of a pressure differential to overcome these relatively small resistances. It is evident from the above

discussion that any reduction in pressure at the suction of the pump would result in the flashing of the liquid Freon to the gaseous state. This type of flashing results in a reduction of pump capacity and a new equilibrium results with the new and slower circulation rate.

No matter which type of pump is selected, it will tend to have a reduced pressure zone at the pump suction. This pressure reduction zone can be controlled somewhat by slower pump speeds, proper piping configuration, etc. but cannot be eliminated entirely. A sufficient column of liquid above the pump can overcome this low-pressure zone. Columns of this type, commonly referred to as a "suction head" must produce enough additional pressure at the suction of the pump from the hydrostatic forces produced by gravity to overcome the low-pressure zone for proper pumping operation. A suction head of this type does exist in the ice tank but is constantly changing because of the constantly changing load on the evaporative coil resulting in changing pressures on the condensing coil. The amount of liquid in the condensing coil changes constantly because of the constant change in load on the evaporative coil. As a result, this type of hydrostatic head is unreliable and can't be counted upon to be consistently stable.

A further complication to the system is the viscosity of the liquid Freon. At the condensing pressure and temperature, the Freon viscosity is only about 15% that of water. Any increased backpressure on the pump discharge results in slippage of the liquid back through the pump from the discharge to the suction. This slippage greatly affects the capacity of the pump and causes eddy currents in the fluid. These eddy currents cause additional low pressure zones that cause additional flashing of the liquid to its gaseous state. Traditionally, designers have utilized increased speed of the pump, tighter clearances of the internal pump parts, and special designs of the pump to control this slippage. However, these attempted solutions cause the pump price to increase dramatically and the problems are not eliminated but merely reduced. The low viscosity also affects the required seal design of the pump. Generally speaking, a sealless pump design is preferred in order to avoid problems, such as a magnetically driven pump. The requirement of a magnetically driven pump also increases the cost of the installation, however.

When the expansion valve is considered on the discharge of the pump, a back pressure of over 100 psi is induced. This back pressure is considerately higher than any pressure drops associated with line

sizes or with respect to the pressure considered with elevation changes. Slippage and low pressure zones increase dramatically, resulting in system failure. In order to insure system integrity, a pump design must be provided which effectively prevents any slippage.

At the condensing temperature in the ice tank, the viscosity of the liquid Freon is only about 15% that of water at the same temperature (0.553 lb/ft-h compared to 3.690 lb/ft-h). This extremely low viscosity causes a variety of pump problems, especially when trying to pump the liquid to significantly higher pressures. These problems include extreme slippage and the need for expensive shaft seal designs negatively affecting the pumping efficiency and pump cost.

These problems can easily visualized by referring to Figure 1 of the drawings. Because of the need to develop a significant pressure differential across the pump, a positive displacement pump is preferred. In Figure 1, a gear pump is shown, although the same problems will apply to any pump, whether or not the pump is a positive displacement pump. The particular pump illustrated in Figure 1 has a housing 21 that contains a chamber that has a driven gear 25, driven by a shaft 26, through a seal and bearing (not shown). This driven gear 25 meshes and turns a mating idler gear 22 that rotates on a shaft 10 resting in a bearing 23. In the drawing, the driven gear 25 is rotated clockwise and the idle gear 22 is rotated counter clockwise. With this rotational direction, a low-pressure zone is developed at an intake zone 18. The liquid the is forced to flow around the gears (clockwise around the driven gear 25 and counterclockwise around the idler gear 22).

If the viscosity of the liquid is high enough, and if the clearance between the gears and the housing is small enough, a large volume of liquid is transferred from the low pressure zone 18 to the opposite side of the housing 21 to a discharge zone 20 causing an increase of pressure in zone 20 relative to zone 18. The liquid in the high pressure discharge zone 20 is retarded from flowing between the gears 22 and 25 to the low pressure zone 18 because of the meshing of the gears. If the mesh is tight enough and the viscosity is high enough, very little leakage occurs. A tight clearance between the gear faces and the housing will retard flow from the high-pressure zone to the low-pressure zone if the viscosity is high enough. The low-pressure zone 18 is connected to the outside of the housing to serve as the "suction" of the pump through the inlet chamber 24. The high-pressure zone 20 is connected to the outside of the chamber with discharge chamber 27 serving as the pump "discharge".

Again with reference to Figure 1, if the fluid being pumped has a very low viscosity, an excess of liquid volume will tend to flow from the high-pressure zone 20 to the low pressure zone 18. This reverse flow of fluid is known as "slippage". Slippage requires that additional speed be employed to turn the gears 22, 25 in order to maintain the desired volume flow rate needed to develop a given pressure differential. Lower viscosity fluids being pumped, higher pressure differentials and greater clearances between the gears all increase slippage. Another problem arises due to the natural lubrication properties of the fluid being pumped since the fluid being pumped normally provides lubrication for operation of the pump. A low viscosity liquid "washes" any lubrication properties of the liquid away from the rotating gear surfaces. This action results in accelerated wear of the bearings and seals utilized in the pump mechanism.

The lowest system pressure of the liquid being pumped occurs at the low-pressure zone 18. If the system pressure was lower anywhere else in the system, the liquid would flow to that zone instead of flowing to the pump suction. If this low-pressure zone falls below the liquid's vapor pressure at operating temperature, vaporization of the liquid will occur. This transformation of the liquid to its vapor state is known as "cavitation" and greatly reduces the volume flow rate of the pump. Experience teaches that cavitation causes excessive pump wear, increases the power needed to achieve volume flow required volume flow rates, as well as producing unpleasant noises.

It can easily be shown that the presence of low viscosity fluids coupled with poor lubrication of the pump gears results in increased leakage around the drive shaft seal. When pumping a liquid that is destructive to the environment leakage of this seal is unacceptable. Because of the low viscosity and lubrication properties of water, this type of pump has been found to be unfit for use with water in most circumstances, resulting in increased power use and premature mechanical failure. Trying to pump a liquid with a viscosity much lower than water predictably results in system failure.

Problems Associated With Flow Control Of The Freon Refrigerant:

Another problem associated with the third type of thermal energy storage system described above can be seen with reference to Figure 3 of the drawings. Figure 3 is a simplified schematic which is intended to represent a basic refrigeration system of the type found in most presently existing

residential and small commercial building structures. The particular structure to be cooled is indicated generally as 31. Inside structure31 is a coil type evaporator, referred to as 32. Evaporator coil 32 is usually made up of several coils that operate in parallel, as illustrated in the drawing. Freon flow entering the evaporator coil 32 is split into equal flows through each of the parallel coils in order to achieve proper heat exchange efficiency. An air blower 34 is provided to move internal structure supply source air across the evaporator coil 32 with heat from the moving air being transferred to the Freon in the coils 32.

A large diameter conduit 35 carries heat laden Freon vapor through the structure wall and outside to the condensing unit 36. The condensing unit 36 consists of a compressor 37 that compresses the Freon vapor to an elevated pressure, a condensing coil 38 that allows the high pressure Freon vapor to change to a liquid state as it loses heat, and an air blower 39 that moves outside air past the condensing coil to carry the heat away from the Freon that was gained inside the structure and was added during compression.

A small diameter conduit 30 carries the high-pressure liquid Freon back through the structure wall to an expansion device 40. The expansion device 40 used in most existing installations consists of two essential components. The first is a metering orifice 44 that has a fixed orifice size to restrict Freon flow. The second essential component of the expansion device is the presence of one very small tube 46 leading from the metering orifice 44 to each coil in the evaporator unit. The purpose of these small tubes is to split the Freon flow evenly into each coil section. This system has proven to be the most economical system that can be installed at the present time that also yields dependable service. As a result it is the most common system in use.

It should also be noted that the metering orifice 44 and the small tubes 46 make up a performing unit which must meet the capacity of the condensing unit with small changes being made by changing the orifice size of the metering orifice 44. Although the above described system offers cost advantages, there is a slow equalization of the Freon pressure across the expansion device 40 when the compressor is not running. This pressure equalization is sometimes referred to as "bleed over" and allows the compressor to restart without a large load across it, thereby reducing the compressor load on the compressor drive motor and allowing the use of the smallest feasible motor.

A disadvantage of the above described existing system is the critical need to have the proper amount of Freon in the system at all times. The amount of Freon in the system, referred to in the industry as the "charge", is critical down to the one-half ounce. Overcharging or undercharging the system will greatly affect operation. An overcharge causes a high head pressure because excess liquid accumulates in the condenser. This accumulation will disable that portion of the condenser coil, thereby increasing the load on the remaining parts of the coil. This raises the temperature of the Freon in the condenser coil and raises the condensing pressure. An undercharge of Freon results in low head pressure because the condenser is, in effect, oversized. Since not enough liquid is being pushed through the expansion device and a low head pressure causes low suction pressure. Vapor tends to bubble in the expansion device because the bubbles further increase flow resistance.

A more expensive and thus less commonly used conventional air conditioning system is illustrated in Figure 4. In the system shown in Figure 4, a thermostatic expansion valve 54 is substituted for the metering orifice of Figure 3. This substitution eliminates most of the problems associated with the requirements for a critical Freon charge. The thermostatic expansion valve 54 opens and closes depending upon the exiting evaporator Freon temperature. A temperature-sensing bulb 51 mounted on the discharge line 55 of the evaporator 50 controls the thermostatic expansion valve 54. With the expansion valve 54 in place, a liquid storage tank or "receiver" 52 can be used between the condensing unit 556 and the expansion device 50. As a result of the presence of these additional components, the Freon charge is not as critical as in the system of Figure 3.

Although the system of Figure 4 is more expensive than the system of Figure 3, it is more versatile under changing load conditions. One disadvantage of the system of Figure 4 is that bleed over cannot be used, as described with respect to Figure 3, because of the possibility of the suction line filling with liquid rather than vapor due to the presence of excess Freon stored in the receiver 52. Liquid of this type present in the compressor suction could result in the destruction of the compressor since it cannot compress liquids. Without bleed over, a larger start up load is present on the compressor, resulting in more expense being required in designing an acceptable compressor drive.

Because of the changes in the Freon system charge from one time period to the next, all of the prior art systems described in the background discussion above utilized the thermostatic expansion valve

54 (Figure 4) to replace the metering orifice (44 in Figure 3). Because of the need to make this and other replacements, the addition of-thermal storage to most residential and small commercial systems requires equipment changes inside the building structure as a part of the retrofit installation.

Because of the above described pumping problems, all the prior art systems described in the background discussion except for U.S. Patent No. 5, 682, 752 (Dean) bypass the thermostatic expansion valve during use of the energy storage tank. Although the '752 Dean patent doesn't address the above described pumping problems, the metering orifice is replaced with a thermostatic expansion valve which is not bypassed. If this described method manages to pump the liquid to the required pressure and volume flow rate, change to the evaporative coil and change to the compressor drive limit the method to new systems rather than a simple addition to an existing system.

Description of the Invention:

The thermal storage system of the invention can be added to an existing Freon compression air-conditioning system such as is commonly found in residential and small commercial structures. These types of systems were described with respect to Figure 3 above. A single retrofit package totally simulates the operation of the conventional condensing unit, using only energy that was previously stored, without any change or alteration of the existing equipment. The method and device of the invention create a new system that stores energy in a first time period, uses this stored energy during a second time period to cool the structure, and possesses the capacity to be able to cool the structure in a third time period without stored energy. Two distinct problems are solved to accomplish these objectives of the invention as described in the discussion which follows.

The Pumping Solution:

The device and method of the invention create a "second time period" when the system uses stored energy to cool the associated building structure. When the second time period arrives in the new system, Freon that was condensed from the vapor state to the liquid state is required to be pumped at a pressure that will be high enough to simulate the running of the compressor in time period three described above. With R-22 Freon (the normally used refrigerant) this pressure has been defined to

be above 200 psi by the air conditioning industry. This pressure has been defined for operation when the outside air temperature is too low to maintain the condensing pressure required to maintain proper flow rates through the expansion device. In the new system of the invention, this pressure can be lower or higher but is used only as a guide for purposes of the present discussion. The problems associated with pumping liquid Freon have been described in some detail. The characteristic properties of the liquid Freon, at the point in the refrigeration cycle where pumping is required, are such that the liquid Freon is difficult to reliably pump. Applicant has discovered that the difficulties associated with pumping liquid Freon at this point in the refrigeration cycle can be overcome by pumping an auxiliary liquid that has the proper properties for pump operation and by then mechanically coupling or transferring the pressure and flow rate of the pumped auxiliary liquid to the liquid Freon.

The auxiliary liquid which is selected will have a much higher viscosity than the conventional liquid Freon present in the system. The auxiliary liquid will also have a lower vaporization pressure than the liquid Freon, will have the required lubrication properties, and will not be dangerous to the environment should it leak around the drive shaft of the pump. Several liquids have such properties and can be used with conventional lubricating oil being one example for purposes of the present discussion. Pumping such an auxiliary liquid results in less energy use because of reduced slippage in the pump, increased longevity of the pump because of its lubrication properties and reduced cavitation.

The mechanical transfer or coupling of the pump pressure and volume flow rate from the pumped auxiliary liquid to the liquid Freon can be accomplished by several methods. The preferred methods discussed below are intended to be illustrative of the principles of the invention without being limiting.

Figure 5 is a simplified schematic of one pumping system which can be used in the practice of the invention. In the device and system of Figure 5, two cylinders 59, 63 each contain a piston 64, 65. The lower face 61, 62 of each piston 64, 65 is coupled a port of a pump 67 by means of a conduit 66, 68, respectively. The pumpable auxiliary liquid is contained in each cylinder below the pistons in a chamber, such as chamber 70, in the pump conduits 66, 68, and in the pump 67. Dynamic seals on the pistons 64, 65 contain the pumpable auxiliary liquid as the pistons move up and down. The upper

or top faces 72, 73 of the pistons 64, 65 in the cylinders are exposed to chambers 66, 67 and communicate with check valve arrangements 69, 70 that allow low pressure Freon in the pump system entrance conduit 71 to enter the cylinder when the piston is moving down and at the same time prevent high pressure Freon from entering the cylinder. When the piston is moving up, the valve arrangements reverse causing the low pressure Freon entrance 71 to be blocked. and the exiting high pressure Freon to flow to the pump system exit conduit 75. The valve arrangements are shown using check valves that allow flow in one direction only. Other valves could be used including individual electrical or pneumatically operated valves, four way valves, shuttle valves, etc..

Because the effective area of the top surface 72, 73 of each piston is the same as the bottom surface 61, 62, the pressure of the liquid above the piston is the same as the liquid below the piston with the exception of any drag that is associated with the dynamic movement of the piston seals (which is small with proper design). Because the exposed piston area are effectively identical, a change in liquid volume below the piston causes the piston to move creating an equal change in volume above the piston. By the same reasoning, if pistons are selected such that the pistons in each cylinder have different effective areas, the pumped pressure will be different than the Freon pressure and a different volume change will result. This might be considered an advantage in some circumstances of pump design. Also, while a gear pump 67 is illustrated, any of a variety of pump designs could be utilized, as will be appreciated by those skilled in the relevant arts.

In the particular embodiment of the pump illustrated in Figure 5, with the pump turning in the direction shown, the left piston 65 is traveling up while the right piston 64 is traveling down. As a result of the described pump direction and the resulting piston movement, low-pressure Freon is entering the top of the right cylinder chamber 66 and the high-pressure Freon is exiting the left cylinder chamber 70. In this illustration, the pump 67 reverses direction when the piston 64 in the right cylinder 66 approaches the bottom of the cylinder chamber 66. The location of piston 64 could conveniently be sensed by a transducer (not shown) that senses the piston location and which operates magnetically, optically, or mechanically, or by sensing the pressure of the pumpable liquid between the piston and the pump. When the pump reverses its rotation, the direction of the piston movement reverses causing low-pressure Freon to enter the left cylinder and high-pressure Freon to exit the right cylinder. By selecting the proper design criteria, the volume of the pumpable liquid in the two

cylinders, the pump, and pump lines can be designed such that when one piston is at the bottom of its stroke, the piston in the other respective cylinder would be at the top of its stroke. Pump capacity will be determined by selecting the diameter and the length of the cylinders, the speed of each piston in its movement within the respective cylinders, and the duration of each pump rotation.

By purposely providing slow and long relative piston movement and stroke, any low-pressure Freon in its vapor state can enter the cylinders 59, 63 without causing harm since it would quickly change to its liquid state with increasing pressure. Instead of a single pump being used, a pump for each cylinder with a volume tank could also be used. This type pumping system has been used by Applicant on a three-ton air conditioning system with the pump consuming less than 1/2 horsepower with a cycle time of about 40 seconds. Experiments have shown that the pump consumes 1/2 horsepower when it attempts to pump liquid Freon to the same pressure and same volume flow rate. The greatly reduced power consumption is due to the slippage and cavitation problems described above.

The reduced power consumption which is achieved in the above example is important for purposes of the invention because it allows, for example, a battery (150 in Figure 8) to be used as a power source for the pump rather than a conventional AC power outlet. A wet cell battery, for example, represents a much smaller capacity and much less recharging energy, allowing it to be recharged by solar panels, or the like. Because of the lubrication properties of the pumped liquid and the absence of cavitation, the life of the pump of the invention, as well as the pumping cost, are greatly reduced.

Figure 6 is another embodiment of the pump which can be used in the practice of the invention to achieve the necessary energy transfer. In the embodiment of Figure 6, inflatable bladders 82, 84 are substituted in each of the cylinders 86, 88 in place of the pistons described in Figure 5. Each bladder 82, 84 separates the pumpable auxiliary liquid from the Freon while maintaining equal pressure on either side of the Freon. This system works essentially the same as the piston method with the advantage of the elimination of piston drag. Likewise, a number of component substitutions can be visualized, as described with respect to Figure 5.

While the invention has been described with respect to the two embodiments of the pump illustrated in Figure 5 and 6, it is not intended to be thus limited and a number of additional embodiments utilizing the principles of the invention can easily be visualized. For example, another example of pumping energy transfer could be achieved by utilizing an auxiliary liquid which is non-mixable with Freon. Such a non-mixable but otherwise compatible liquid could conceivably be used without the further requirement of pistons or bladders to separate the auxiliary liquid from the Freon.

Another example of pumping energy transfer could be achieved by using one pump and one cylinder in conjunction with a surge chamber providing pressure and volume to the high-pressure Freon side of the system while the pump and pump and cylinder were taking in low pressure Freon.

Another example of pumping energy transfer is illustrated in Figure 7 in which the pumping energy transfer system is insulated to prevent additional vaporization of the liquid Freon as a result of the Freon gaining heat. In the system illustrated in Figure 7, the energy transfer cylinders 92, 94 are located within, or at least partly within, the energy storage tank 96.

The Freon Flow Control Solution:

The control of the flow of Freon through the three time frames described above is a critical component of the successful working of the method and device of the invention. In the first time frame, the condensing unit is running and freezing the liquid in the energy storage tank. In the second time frame, the condensing unit is not running and cooling inside the structure is being provided by the operation of the newly added thermal energy storage unit. In a third time frame, the condensing unit is running to cool the structure as if the new thermal energy storage unit were not present. For the energy storage unit to be added to an existing unit without any change to the existing unit, the added unit must allow the operation of all three described time frame operations in transparent fashion. For this type transparent operation to be achieved, it is necessary that no liquid Freon be allowed to accumulate in the condensing coil of the condensing unit as a result of excess Freon in the system. Any accumulation of liquid Freon in the condensing coil would result in an elevated head pressure as has previously been described.

Another important feature for the proper operation of the system of the invention is to preserve the previously described bleed over advantage offered by the metering orifice expansion device. This advantage should be preserved in order to avoid the possibility of loading the evaporator coil with liquid Freon. Such loading could very likely cause damage to the compressor upon start up. The control of the Freon flow in the system to achieve these goals is described by viewing Figure 8.

Figure 8 shows a thermal energy storage system of the invention which can be added to an existing Freon compression air-conditioning system without any major change to the existing system. The system of the invention illustrated in Figure 8 includes an insulated storage tank 132 that houses an energy storage coil 134. The system further includes a liquid Freon pump system 67 as described with respect to Figure 5. The pump system 67 is powered by a battery 150 that is recharged during time period one operations. The system also includes a Freon liquid storage device or "receiver" 143 of the type commonly used in the air-conditioning industry, as well as volume tank called an expansion tank 144. Also shown is an expansion device 135 for the energy storage coil as well as a heat exchanger 136. Seven electrically operated control valves (illustrated as 133, 137, 138, 139, 140, 142 and 145 in Figure 8) are utilized. The unit is installed on the existing air conditioning system by attaching line 148 to the large suction line present in the existing system. The small liquid line of the existing system is interrupted and attached to lines 146 and 147 of the energy storage unit with line146 being attached to the section that leads to the condensing unit and line 147 being attached to the section going inside the structure to the existing expansion device.

The operation of the thermal energy storage system of Figure 8 will now be described. The receiver 143 is a pressure-containing vessel that has one opening 154 in the top thereof that receives liquid Freon. A line 156 exits the top of the receiver 143 and also extends in close proximity to the bottom surface 158 of the vessel to allow exiting of the liquid Freon. Freon is taken from the bottom of the vessel 143 in order to insure that liquid Freon leaves the vessel, if the vessel contains any liquid. The expansion tank 144 is a pressure containing vessel that simply provides volume for expansion of any liquids that might "slug" into the compression suction line. A conventional absorber has long had common use in the industry and could be used.

The insulated storage tank 132 contains a liquid storage medium that stores cooling energy. In the preferred case, this medium is water. In the first time period heat is removed from the liquid through the coil134 housed within the tank 132. In most cases heat is removed until the liquid changes state to its solid form. If the liquid being utilized is fresh water, the heat removal required for changing phase to ice would be 144 BTU per pound of water after the liquid dropped to 32 degrees F. In the second time period heat is added to the solid until it changes state back to a liquid. This heat is added through the same coil 134 that extracted the heat in the first time period. Each pound of ice would theoretically absorb 144 BTU of heat before changing phase back to water. The heat exchanger 136 simply pre-cools the warm liquid Freon going to the expansion device 135 with cool Freon vapor leaving the coil before it returns to the compressor. The expansion device could be of various types, but a thermal expansion valve as described above would provide the best service. The control valves are either in the open or closed position depending upon the control signal going to the valve.

TIME PERIOD ONE

With valves 137, 140, 142 and 145 in their open position and the remaining valves closed, the unit would be in the energy storage mode. Liquid Freon under pressure would be entering the unit through line 146 from the condensing unit, passing through valve 145. The liquid would enter the receiver 143 and exit as a liquid. The liquid would then travel through valves 142 and 140 through the heat exchanger 136 to the expansion device. Flow through the expansion device would result in a much lower pressure upon passage through the device. The Freon would change states to its vapor state as it passes through the low-pressure area and would simultaneously extract heat from the liquid surrounding the coil 134. The vaporized Freon would travel through the heat exchanger 136, valve 137, and then exit the unit through line 148. The line 148 returns the Freon to the condensing unit where it is compressed to a hot high pressure vapor and then condensed to a high pressure liquid in the condensing coil. This process would continue for the duration of time period one or until all the liquid in the storage tank changed to its solid state, at which time the condensing unit would stop running. More Freon is in the system than is required for this mode of operation and collects in the receiver. The pump battery 150 is fully charged during this time period.

TIME PERIOD TWO

This time period requires the structure to be cooled with energy that was stored during time period one. As will be explained, this requires utilization of all of the Freon contained in the system. This mode is operated with valves 133, 137 and 139 in the open position. The pump system is in operation as described above. All the liquid Freon in the receiver 143 leaves the high pressure warm receiver through valve 133 to the suction of the pumping system at a faster rate than it can be pumped. The excess flow goes into the energy storage coil 134 where it provides the required net positive suction head for the pumping system. After a sufficient time for the receiver 143 to empty, valve 133 closes. The pumping system pumps the liquid to a pressure and flow rate to simulate the running of the condensing unit. The liquid leaves the pump system 67 and passes through valve 139 and leaves the unit through line 147. The liquid then travels inside the structure to the expansion device 135 with the same flow rate and pressure as it would have as if the compressor were running.

The Freon then vaporizes in the evaporator coils 134 gaining heat from the structure. It then travels through the existing line through the structure and then into line 148 where it passes through valve 137 into coil 134. When the Freon enters the coil 134 submerged in the cool liquid, it condenses to its liquid state and passes the heat gained inside the structure to the cool liquid. After the Freon condenses to its liquid state, it enters the suction of the pumping system. As long as the pump system is running and the liquid in the energy storage coil condenses the Freon to its liquid state, the second time period exists. The pumping system is turned on and off in the same manner as the compressor would be turned on and off if it were in operation. This action controls the temperature of the air inside the structure the same as if the compressor were active. All of the Freon is in circulation in this time period. At the end of this time period or in the event the energy storage properties of the liquid in the energy storage tank is depleted, the bulk of the Freon is transferred back into the receiver.

This last action is accomplished by an industry standard method called a "pump down" cycle in which all of the valves except valves 137 and 145 are closed and the condensing unit is turned on. Freon is pulled into the compressor as a vapor and compressed. The compressed Freon flows into the receiver 143 where pressure is increased until it liquefies. Any liquid that might flow into line 148 during the pump down cycle falls into the expansion tank 144 where it expands to its vapor state, thus

protecting the compressor. This continues until the pressure inside the energy storage coil decreases to a preset level (that corresponds to a temperature much lower than it would be if the storage liquid were in its solid state). A pressure switch (not shown) senses this pressure. When the pressure is decreased to the preset level, the compressor is turned off and valves 137 and 145 are closed.

THIRD TIME PERIOD

This time period requires the condensing unit to run and transfer heat from the structure to the outside as if the energy storage unit were not present. Time period three operations occur if cooling is required inside the structure during the first time period or if the energy storage properties are depleted during time period two. This is accomplished by the compressor coming on as normal and valves 139, 142 and 145 being in the open position with the remaining valves being closed. The liquid Freon from the condensing unit flows into the new system through line 146, through valve 145, and into and out of the receiver 143. Liquid Freon then flows through valves 142 and 139 to the inside of the structure to the expansion device. Then vaporizing Freon in the evaporator coils gains heat and then travels to the compressor suction where it is compressed and then condensed to a liquid in the condensing coil. The Freon then flows back to line 146. This is the same path with the same pressures and flow rates that the Freon would be taking and would encounter if the new energy storage device were not present with the exception of flowing through the receiver.

The purpose of receiver 143 in this time period is the prevention of the excess Freon in the system from collecting in the condenser coil where it would cause increased head pressure. Excess Freon is accumulated in the condenser without affecting the operation of the system. When the compressor goes off because the air becomes cooled inside the structure bleed over would normally occur as described above. This is achieved by the opening of valve 138 in conjunction with closing of valves 139, 142 and 145. When these valves shift, the suction and discharge of the compressor is equalized and is restarted with the same ease as before the energy storage device was installed. These valves stay in this position until the compressor comes back on, at which point they return to normal third period operation.

Figure 9 illustrates the unit described in Figure 8 but with the unit installed in the existing system as described in Figure 3. It should also be noted that, if the existing system were the more expensive system containing a thermal controlled expansion valve and a compressor not requiring bleed over, the thermal energy system of the invention could be utilized in that type system as well with similar results.

An invention has been provided with several advantages. The thermal energy storage system of the invention can easily be added to an existing Freon compression air-conditioning system commonly used in residential and small commercial applications. The system of the invention is also easily incorporated into new construction with relatively little additional expense. The method and device of the invention allow the addition of a single "package" to a new or existing air conditioning system that totally simulates the operation of the conventional condensing unit. The system of the invention uses only energy that was previously stored by the system and does not require any change or alteration of existing equipment inside the structure. The add-on package of the invention creates a new system that stores energy in a first time period, utilizes this stored energy during a second time period to cool the structure, and possesses the capacity to be able to cool the structure in a third time period without utilizing stored energy.

In addition to providing an add-on package which can be easily installed outside the structure, two distinct problems are solved by the improved system of the invention. The problem of pumping a low viscosity and easily vaporized liquid such as Freon is solved. Additionally, the problem of controlling the Freon flow and charge in each time period is solved without dependence upon the particular type of system which existed initially and was retrofitted.